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# Research and Solution of Vibration Problem of an Air-cooled heat exchanger of Reciprocating Compressor

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# Research and Solution of Vibration Problem of an Air-cooled heat exchanger of Reciprocating Compressor

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**Abstract:** Excessive vibration of complex structures in a multi-excitation environment is an arduous problem. Especially in a reciprocating compressor, how to quickly and accurately position the vibration sources, analyze the excitation features and take effective solutions is of great significance to the safe and stable operation of the compressor. In this paper, the problem of excessive vibration of an air-cooled heat exchanger used in a variable speed CH4 compressor was studied. The air-cooled heat exchanger is composed of tubes, tube boxes and support frame. Its vibration is evoked by complex periodic gas pulsation in the tubes. The vibration model of the heat exchanger was simplified as a multi-freedom model with periodic excitations using the lumped parameter method. The model has advantages of clear physical concept, easy solution and high accuracy of result. In the model, the air-cooled heat exchanger was simplifies as three mass blocks connected by six springs. And then, modal test of the heat exchanger was carried out on site. The vibration modes and natural frequencies of the exchanger were obtained according to the test results. After that, above mentioned multi-freedom model was further simplified as a single freedom vibration model, so that it was convenient to propose the vibration control measures. The practical application results proved that these measures were effective. This research provided a practical and fast analytical method for solutions of compressor vibration problems.

**Key words:** air-cooled heat exchanger, complex structural vibration, modal test, lumped parameter method

## 0 Introduction

Reciprocating compressor are core equipment for natural gas gathering and transportation. The inter-stage cooling and post-cooling of the compressor are air-cooled or water-cooled heat exchangers. Among them, air-cooled heat exchangers are favored by most manufacturers and users of reciprocating compressors because of simple structure, convenient design and manufacturing, and no additional water consumption. The periodic suction and exhaust of gas will cause the vibration, fatigue and even cracking of the air-cooled heat

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exchanger, thus greatly threatens the stability and safety of the compressor. Especially for large-scale air-cooled heat exchangers, due to their large mass and large span of tube box support, the vibration problem is more prominent and more serious. Therefore, it is very important to analyze and test the vibration of the air-cooled heat exchanger and take anti-vibration measures to ensure the safe operation of the compressor.

In order to describe the dynamic response of air-cooled heat exchanger, various mathematical models have been established,such as lumped parameter model, multi-body dynamics model, and finite element model are the most general models.The lumped parameter model is used for one-way analysis,it can extract obvious vibration characteristics from complex structural parts to establish a model, and ignore other minor or insignificant characteristics.It’s widely used in practical engineering problems.

The difficulty to establish a lumped parameter model is how to effectively and accurately calculate the lumped parameter. Generally,people are more likely to use methods of the 3D model analysis [1~2] and modal test analysis [3~5]. This paper proposed a new method based on the combination of modal test [6] and finite element static analysis to slove an engineering problem, greatly accelerate the calculation speed. After field test , comparison and verification, this method was proved to be good.

**1 Descriptions of the air-cooled heat exchanger vibration problem**

The compressor is composed of three rows with two stage compression, and key parameters are shown in Table 1. At 300 rpm or so, severe vibration happened at the heat exchanger, while it operates normally at other operating speeds. As shown in Figure 1, the air-cooled heat exchangeris mainly composed of three parts from the bottom to the top, i.e. heat exchanger of the second stage exhaust gas, of the first stage exhaust gas, and of cooling water of the engine, respectively. Vibration mainly occurs in the direction of tube length with a maximum value of 36mm/s (RMS) (2mm (RMS)) at the tube box, which is far over the limits in standard of ISO 13706 " Petroleum, petrochemical and natural gas industries - Air-cooled heat exchangers ". After excluding the possibility of vibration of the compressor and the engine causing the vibration of the air-cooled heat exchanger, we focused on reasons of excessive gas pulsation in pipeline and structure resonance.

Table 1 Compressor parameters

| Parameters | Values | Parameters | Values |
|---|---|---|---|
| Suction pressure/MPa | 0.7 ~ 3.0 | Stroke /mm | 280 |
| Suction temperature /℃ | -5 ~ 40 | Operating speeds r/min | 300 ~ 400 |
| Discharge pressure/MPa | 6.3 | Volume flow rate ×104 / Nm3/d | 10.7-56.1 |
| Number of stages | 2 | Driver | Natural gas engine |
| Number of first stage cylinders | 2 | Driver power /kW | 630 @ 440RPM |
| Number of second stage cylinder | 1 | | |

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Fig.1 Size of the air-cooled heat exchanger

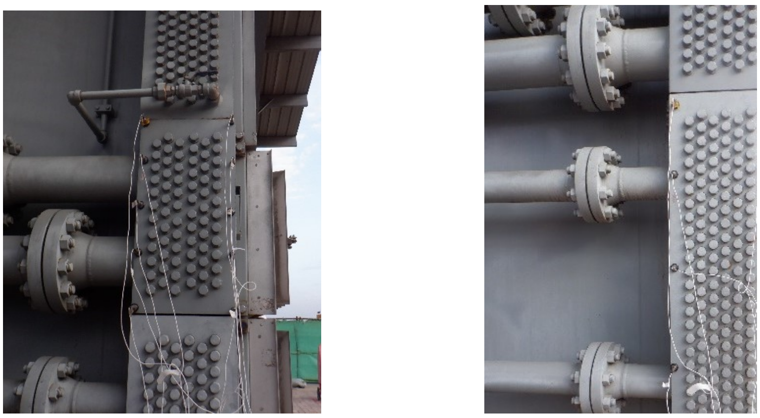
## 2 Vibration tests of the air-cooled heat exchanger

Between 300~400 rpm, we tested the vibration of the heat exchanger[7], then analyze the frequency spectrum of vibration signals. The analysis results show that, between 300~330 rpm, there was an obvious peak around 11Hz (maximum about 60m/s^2); between 340~400 rpm, the peak value at this frequency became smaller and smaller (minimum about 1m/s^2), then gradually became equal to the harmonics at other frequencies at the Z-direction (tube bundle length direction).

At the same time, we tested the pulsation in the pipe neer the heat exchanger, analyze frequency spectrum of pulsation signals. The analysis results show that, between 300~400 rpm, the peak values of the pulsation were within the API618 standard ranges[8] (maximum was about 20 KPa). The frequency multiplication components are very obvious at each multiple speed, and decrease in turn.The amplitudes of harmonics greater than 60Hz are small(less than 1Kpa) ,which can be ignored.

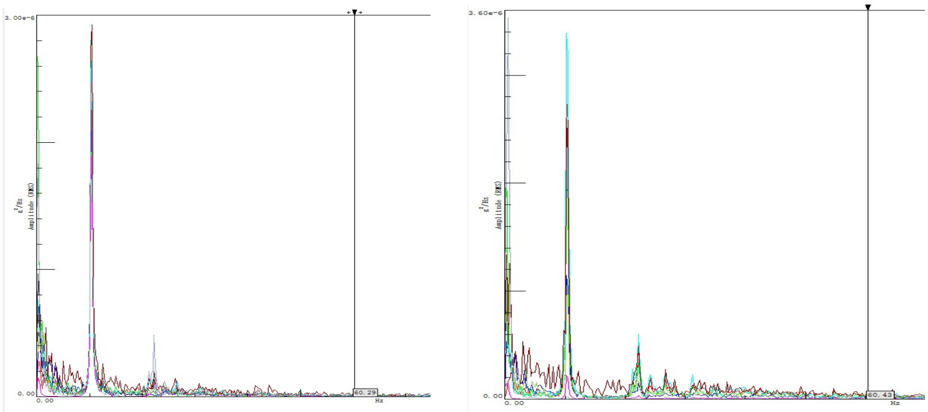
According to the test results above, it was speculated that vibration resonance of the exchanger had occured,so the modal test of the exchanger was proposed.

Since the vibration mainly occurs in Z direction, the modal test was only conducted in Z direction, and sensors were set on the middle and lower part of the head, which are shown in Figure 2. The node diagram of modal analysis was established in the LMS Test.lab system.



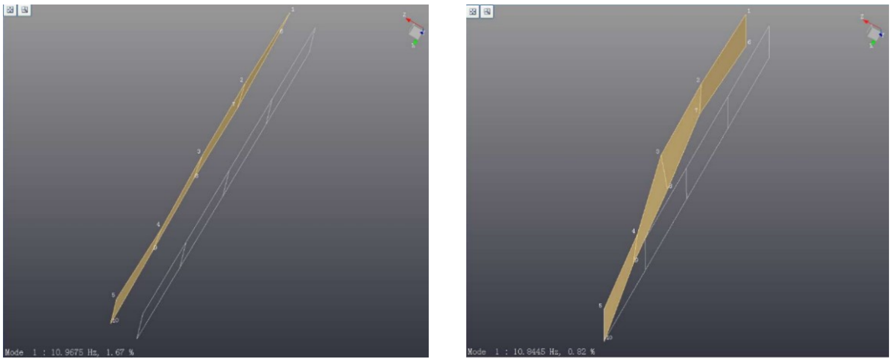
(a) Layout of sensors on primary tube box (b) Layout of sensors on secondary tube box

Fig. 2 layout of sensors on tube boxes



(a) Autocorrelation spectrum of primary tube box (b) Autocorrelation spectrum of secondary tube box

Fig. 3 Response spectrum of vibration in z-direction



(a) Mode of primary tube box（10.9675 Hz） (b) Mode of secondary tube box (10.845Hz)

Fig. 4 Vibration mode of the tube boxes

The test results of the autocorrelation spectrum of each sensor are shown in Figure 3, the results tell that a remarkable peak occurs around 11 Hz, the amplitude of the 2nd and 3rd harmonics of the primary and secondary tube box is gradually reduced. At the same time, the higher-order modes will only appear at the frequency over 60 Hz in the spectrogram.As the harmonics greater than 60Hz of the pulsation excitation can be ignored, the higher-order modes can be ignored also. The vibration mode of the exchanger around 11 Hz shown in Fig. 4 were drawn according to the response of sensors, There is only one node in the mode of vibration around 11 Hz,it means that it's the first mode of the exchanger.

From the analyse results above, it can be determined that the reason for the excessive vibration of the heat exchanger is that a first-order resonance happed at exchanger in Z direction at a frequency around 11 Hz, and the the excitation source is the pulsation harmonic around 11Hz(2nd times of fundamental excitation frequency between 300~330 rpm). After test another exchanger in the same way, we got the same conclusion.

**3 Theoretical analysis of Air-cooled heat exchanger Vibration**

The 3D model of the support frame of the air-cooled heat exchanger is shown in Figure 5, and heat exchanger's tube boxes of the secondary exhaust gas, of the primary exhaust gas, and of cooling water are all fixed on the support frame by bolts.

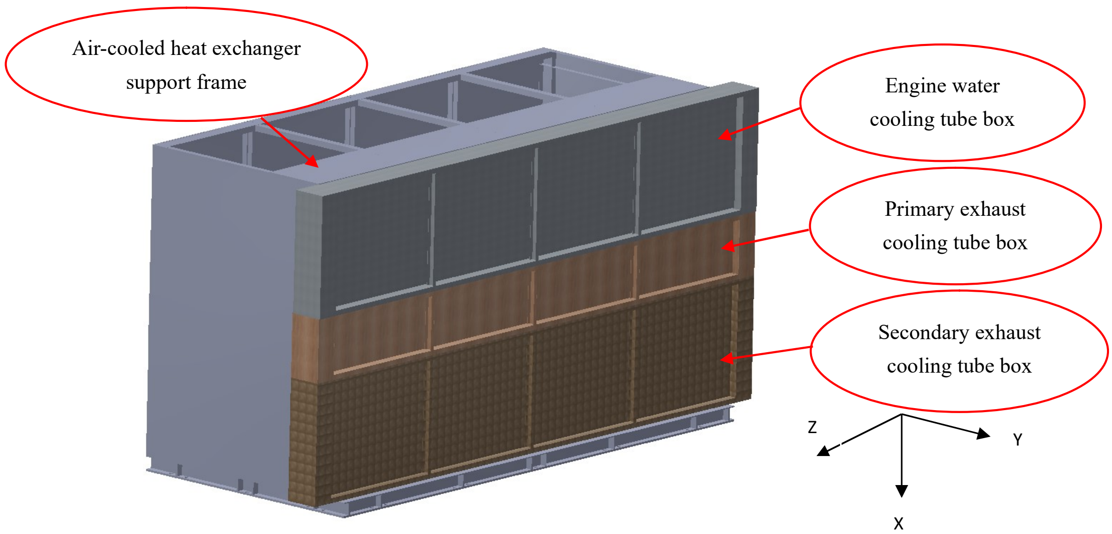



Fig.5 3D model of the air-cooled heat exchanger

Natural frequency of a single freedom vibration system can be calculated by (1) [9]:

$$f = \frac{\omega}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \tag{1}$$

Where f is natural frequency, $\omega$ is circular frequency, K is stiffness ,and M is mass. It's easy to see that f is positively correlated with K , but inversely correlated with M.

The Air-cooled heat exchanger is a welded assembly with complex structures. If modal analysis is conducted according to the real structure such as Fig.5, it will consume a lot of computing resources and time, the fact is it's no need to obtain high level harmonics at all. Therefore, the real air-cooled heat exchanger should be simplified in the modal analysis reasonably. So we ignoring the fan, small accessories, etc., which have negligible effects on the natural frequency of the exchanger, then stablish 3D model of support frame as shown in Figure 6:

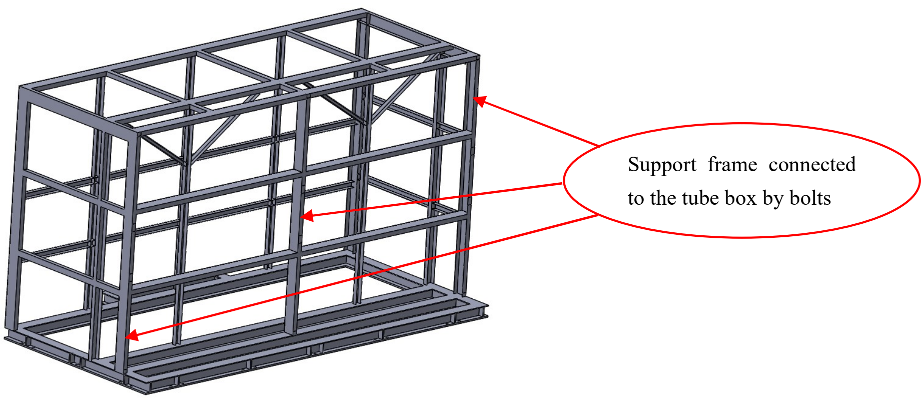



Fig.6 Air-cooled heat exchanger support frame

According to the Air-cooled heat exchanger data, the mass distribution is shown in Table 2:

Table 2 Mass distribution of the heat exchanger

| Serial number | Name | Weight /kg | Media weight /kg |
|---|---|---|---|
| 1 | Engine water cooling tube box | 2,060 | 300 |
| 2 | Primary exhaust cooling tube box | 1,526 | 4 |
| 3 | Secondary exhaust cooling tube box | 2,397 | 3 |

According to Table2, and take three tube boxes as research object, the heat exchanger can be simplified as a multiple freedom vibration system using lumped parameter method,the three tube box can be simplified as three lumped masses without stiffness,the support frame and the connection bolts can be simplified as six springs without mass,establish a multiple freedom model as shown in Figure 7:

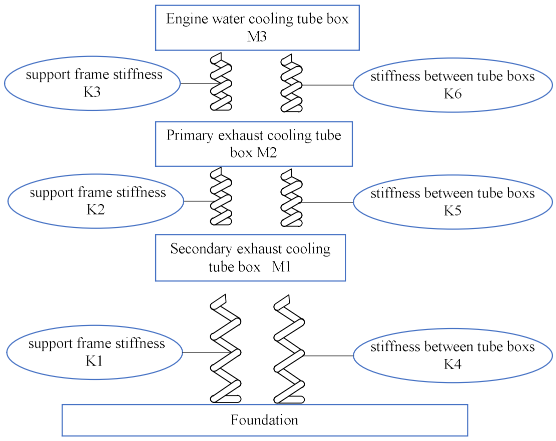
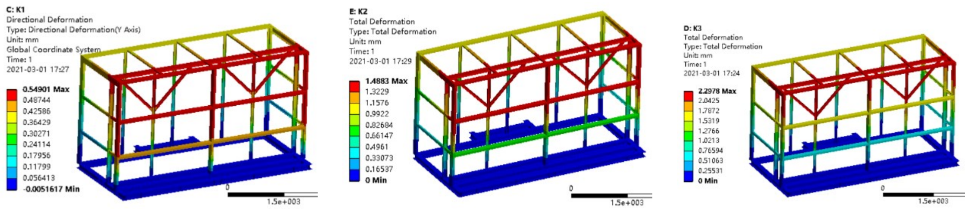



Fig.7 Lumped model of Air-cooled heat exchanger

Calculate each support frame stiffness with FEM, the results are shown in Figure 8[8]:



support frame stiffness K1 support frame stiffness K2 support frame stiffness K3

Fig.8 stiffness calculation of Air-cooled heat exchanger structure

The results of K1, K2, K3 are 7.02E6 N/M,3.12E6 N/M,3.51E6 N/M, respectively. But K5, K6, K7 are difficult to calculate, so they need to be further simplified and equivalent.

The equation of free vibration of a multiple freedom vibration system without damper is as following.

$$M\ddot{x}+Kx=0$$

Where, $M$ is the structural mass matrix, $K$ is the structural stiffness matrix, $x$ is the displacement of each freedom.

Make $x=\phi pe^{i\omega t}$, then multiply both sides by $\phi^T$, and the transformation will be:

$$\omega^2\phi^T M\phi p=\phi^T K\phi p$$

Where, $p$ is response in modal coordinates, $\phi$ is modal coordinates, $e^{i\omega t}$ is complex form of vibration,and t is time.

Assume the mass of the multi-degree-of-freedom system is not coupled, and the stiffness is coupled, the degree of freedom is n, then the formula can be expressed as:

$$\omega^2\phi^T\begin{bmatrix} m_1 & 0 & \dots & 0 \\ 0 & m_2 & & \dots \\ \dots & & \dots & 0 \\ 0 & \dots & 0 & m_n \end{bmatrix}\phi p=\phi^T\begin{bmatrix} k_{11} & k_{12} & \dots & k_{1n} \\ k_{21} & k_{22} & & \\ \dots & & \dots & \dots \\ k_{n1} & k_{12} & \dots & k_{nn} \end{bmatrix}\phi p$$

According to the principle of energy conservation, the maximum kinetic energy and maximum elastic potential energy of each degree of freedom are equal and mutually converted in the free undamped vibration system:

$$E_m = E_k$$

$$E_m = \frac{1}{2}\dot{x}^T M\dot{x} = \frac{1}{2}\phi^T M\phi(\omega p)^2$$

$$E_k = \frac{1}{2}x^T Kx = \frac{1}{2}\phi^T K\phi(p)^2$$

Make the natural frequency of a certain order of the system be $\omega_x$, vibration shape is $\phi_x$ ,and $\phi_x=\begin{bmatrix}\varphi_1 & \varphi_2 & \dots & \varphi_n\end{bmatrix}^T$. According to the orthogonality of the mode shape, the formula can be transformed into:

$$E_m=\frac{1}{2}\omega_x^{\ 2}(m_{11}\varphi_1^{\ 2}+\dots+m_{nn}\varphi_n^{\ 2})p^2$$

$$E_k = \frac{1}{2}(\varphi_1(k_{11}\varphi_1+\dots+k_{1n}\varphi_n)+\varphi_2(k_{21}\varphi_1+\dots+k_{2n}\varphi_n)+\dots\varphi_n(k_{n1}\varphi_1+\dots+k_{nn}\varphi_n))p^2$$

If the kinetic energy of the system is lumped to the first degree of freedom, the formula can be transformed into:

$$E_m = \frac{1}{2}\omega_x^{\ 2}(m_{11}(\frac{\varphi_1}{\varphi_1})^2+\dots+m_{nn}(\frac{\varphi_n}{\varphi_1})^2)\varphi_1^{\ 2}p^2=\frac{1}{2}\omega_x^{\ 2}(m_{11}+\Delta m)\varphi_1^{\ 2}p^2$$

Where $\Delta m=m_{22}(\frac{\varphi_2}{\varphi_1})^2+\dots+m_{nn}(\frac{\varphi_n}{\varphi_1})^2$

So the equivalent mass of M is $M = m_{11}+\Delta m$ (2)

Similarly available, when the potential energy is lumped to the first degree of freedom, the formula above can be transformed into:

$$E_k = \frac{1}{2}((k_{11}(\frac{\varphi_1}{\varphi_1})+\dots+k_{1n}(\frac{\varphi_n}{\varphi_1}))+\dots+\frac{\varphi_n}{\varphi_1}(k_{n1}\frac{\varphi_1}{\varphi_1}+\dots+k_{nn}\frac{\varphi_n}{\varphi_1}))\varphi_1^{\ 2}p^2=\frac{1}{2}(k_{11}+\Delta k)\varphi_1^{\ 2}p^2$$

Where $\Delta k = k_{12}(\frac{\varphi_2}{\varphi_1}) + \ldots + k_{1n}(\frac{\varphi_n}{\varphi_1}) + \ldots + \frac{\varphi_n}{\varphi_1}(k_{n1}\frac{\varphi_1}{\varphi_1} + \ldots + k_{nn}\frac{\varphi_n}{\varphi_1})$

So the equivalent stiffness of K is $K = k_{11} + \Delta k$ (3)

Because $E_m = E_k$,the formula 2 can be transformed into:

$$f = \frac{\omega_x}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{k_{11} + \Delta k}{m_{11} + \Delta m}} \quad (4)$$

Because the structure of the engine water cooling tube box is the same as the Secondary exhaust cooling tube box, the static deformation should be the same. Define the Primary exhaust cooling tube box M2's vibration amplitude as 1, according to Fig 4,the vibration mode of the three tube boxes should be $\phi_x = [1.5, 1, 0.5]^T$.

According to Rayleigh Law and formula 2,take the Primary exhaust cooling tube box M2 as a lumped point, M1, M2, M3 are equivalent to M[9~11]; According to formula 3,K1, K2, K3 are equivalent to D1; K4, K5, and K6 are equivalent to D2; so the total equivalent stiffness should be D=D1+D2. The heat exchanger shown in Figure 7 can be simplified to the equivalent single degree of freedom model shown in Figure 9.

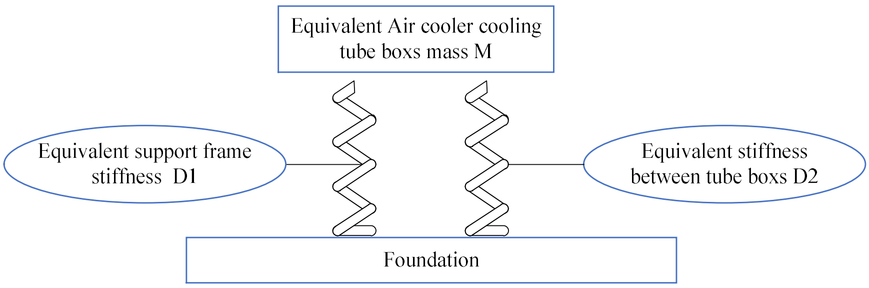



Fig.9 Equivalent lumped model of Air-cooled heat exchanger

According to formula 2, the equivalent mass is M=7440kg, then according to formula 1, the equivalent stiffness of the heat exchanger is D=D1+D2=3.53E7 N/M. According to K1, K2, K3, and formula 3, the equivalent frame stiffness is D1=3.41E6 N/M, so the equivalent stiffness between tube boxes D2=3.19E7 N/M can be obtained.

It is found that the modification of the tube box's mass is very expensive, but the structure stiffness modification will be much easier, so the modification plan is proposed as shown in Figure 10:

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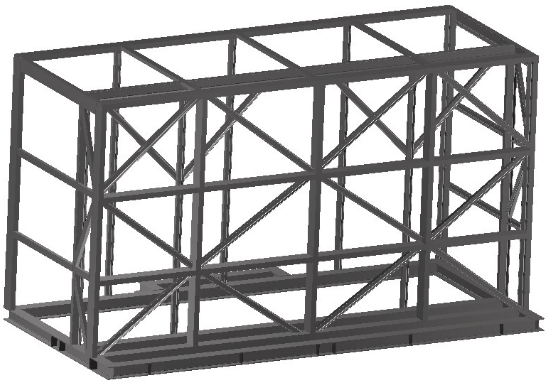
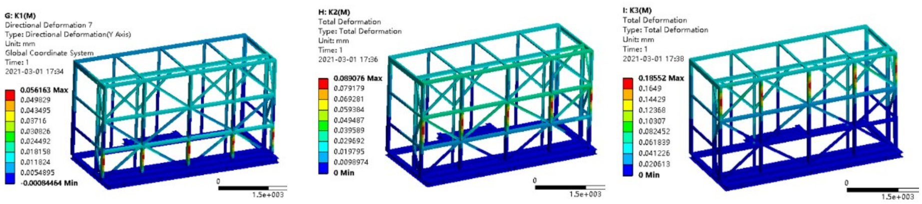


Fig.10 Support frame (modified)

The FEM is also used to calculate the stiffness of the modified air-cooled heat exchanger support frame as shown in Figure 11[12~13]:



support frame stiffness K1 support frame stiffness K2 support frame stiffness K3

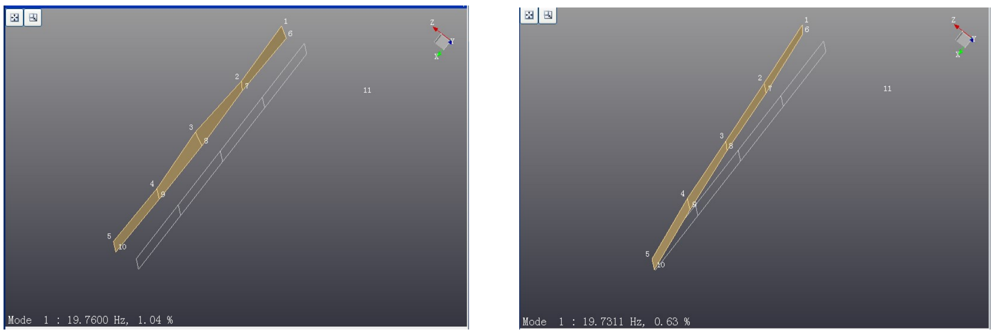
Fig.11 Structural stiffness calculation

After modification, K1, K2, K3 are 1.05E8 N/M, 1.64E8 N/M, and 5.92E7 N/M, respectively.

The first-order mode of the modified Air-cooled heat exchanger is still assumed as $[1.5,1,0.5]^T$, and the welding coefficient is taken as 0.95. According to formula 3, the equivalent stiffness of the modified structure is D1=7.79E7 N/M, and the equivalent mass M and D2 was not changed. So natural frequency of the air-cooled heat exchanger in Z-direction was calculated to be 19.34Hz, which is 2.4 times greater than the highest fundamental excitation frequency required by API618 standard.

## 4 Solution verification

Support frame of the Air-cooled heat exchanger was improved according to above modeling results, and then vibration of the improved air-cooled heat exchanger was re-tested., The test results are shown in Figure 12:




(a) Mode of primary tube box（19.76 Hz） (b) Mode of secondary tube box (19.73 Hz)

Fig. 12 Vibration mode of the tube boxes

The error of first order frequency of calculation result and the test one is: (19.76-19.34)/19.76=2.1%<10%, which meets the API618 standard requirement.

The compressor is restarted under different load. The vibration velocity value (RMS) at tube box under any speed is less than 5mm/s, which is less than the API618 standard requirement. The Air-cooled heat exchanger vibration problem is successfully solved.

## 5 Conclusions

In this paper, the vibration problem of the air-cooled heat exchanger of reciprocating compressor was tested, analyzed and treated. Through on-site testing, it is determined that the cause of the air-cooled heat exchanger vibration is the low-order mechanical resonance caused by the pulsation. Due to the complex structure of the Air-cooled heat exchanger, it is difficult to perform vibration modal analysis based on the real structure of the air-cooled heat exchanger. This paper proposed a lumped parameter method, by which the heat exchanger was simplified a multiple freedom model composing several mass connected by springs. The equivalent mass and equivalent spring stiffness are calculated based on the modal test results. Using this model, improvement design was made for the air-cooled heat exchanger, and vibration of the improved air-cooled heat exchanger was tested. The test results show that the first-order natural frequency of the heat exchanger measured agree well with the calculated one, with a deviation of 4% only. The Air-cooled heat exchanger vibration is reduced from 36mm/s (RMS) to 5 mm/s (RMS). Therefore, the vibration modeling method for complex structures proposed in this paper has good engineering practicability and can effectively solve engineering vibration problems .

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